CHAPTER 3

POWER TRAIN AND PROPULSION SYSTEMS

As a GS supervisor, you will primarily supervise the operation and maintenance of the power train equipment and controllable pitch propeller systems. This chapter will focus on the maintenance and repair of the main propulsion systems needed to support the operations of the main propulsion gas turbine engines.

After studying the information in this chapter, you should have a well-rounded understanding of the drive train equipment and propulsion plant systems in gas turbine-powered ships. You should better understand terms of normal operations, some common malfunctions, and your role as the GS supervisor.

POWER TRAIN

In Gas *Turbine Systems Technician (Electrical) 3/Gas Turbine Systems Technician (Mechanical) 3,* volume 1, NAVEDTRA 10563, there is a fairly detailed description of the various power train arrangements used by gas turbine-powered ships. You may wish to review those chapters on such items as construction, principles of operation, nomenclature, and operating parameters. In this section, we will cover some of the power train system tests, inspections, adjustments, and repairs that you will be responsible for as a supervisor.

MAIN REDUCTION GEAR

The inspection procedures and problems that occur in main reduction gears (MRGs) are basically the same for any system. It will not be necessary to differentiate between classes of ships in this section, except where specific differences exist. Additional information on the inspection and adjustment of gear trains can be found in *Naval Ships' Technical Manual (NSTM),* chapter 9420, "Propulsion Reduction Gears, Couplings, and Associated Components," NAVSEA 0901-LP-420-0002, or in the manufacturer's technical manual for your specific installation.

Inspection and Repair

Before reading descriptions and details on MRG inspections, you need to be familiar with the terminology used throughout this section. The majority of the following gear nomenclature also applies to helical gears. Figure 3-1 may be of help on some of these definitions.

RATIO. The number of gear teeth divided by the number of teeth in the pinion.

LINE OF ACTION. The locus of the points of contact as the profiles go through mesh. This line passes through the pitch point and is tangent to the base circle.

HELIX ANGLE (fig. 3-l). The angle formed by a tooth and a plane passing through the axis of the gear.

PRESSURE ANGLE (fig. 3-1). The angle between the line of action and the line tangent to the pitch circles.

TRANSVERSE DIAMETRAL PITCH. The ratio of the number of teeth to the number of inches of the pitch diameter.

NORMAL DIAMETRAL PITCH. The transverse diametral pitch divided by the cosine of the helix angle.

CHORDAL TOOTH THICKNESS (normal) (fig. 3-1). The thickness of the tooth measured on the chord of the pitch diameter in the normal plane.

CIRCULAR PITCH (axial) (fig. 3-l). The length of the arc on the pitch circle between similar points of adjacent teeth in the plane of rotation.

CIRCULAR PITCH (normal) (fig. 3-l). The length of the arc on the pitch circle between similar points of adjacent teeth in the normal plane.

OUTSIDE DIAMETER (fig. 3-l). The diameter measured over the tops of the teeth.

PITCH DIAMETER (fig. 3-l). The diameter of the pitch circle.

BASE DIAMETER (fig. 3-l). The circle from which a line is unwound to generate the involute curve.

ROOT DIAMETER (fig. 3-1). The diameter of the root circle.

ADDENDUM (fig. 3-1). The distance from the pitch circle to the top of the tooth.

DEDENDUM (fig. 3-1). The distance between the pitch circle and the bottom of the tooth space.

WORKING DEPTH (fig. 3-1). The depth to which the teeth of a gear enter into their mating space.

CLEARANCE (root) (fig. 3-1). The distance between the top of a tooth and the bottom of its mating space.

WHOLE DEPTH. The total depth of the tooth space and also the sum of the addendum and dedendum.

INTERFERENCE. Contact between mating gears at some point other than along the line of action.

FILLET (fig. 3-l). The concave radius that joins the tooth profile and the bottom of the tooth space.

Tests and inspections according to the PMS are minimum requirements only. When defects are suspected, or operating conditions indicate, inspections should be made at more frequent intervals.

When opening gear cases for inspection, use extreme care. Even under normal conditions, when the covers are lifted the possibility of getting foreign particles inside the gear case is high. The engineer officer should evaluate all ongoing work in the engine room, especially in the areas over the gears. Besides evaluating the work, the engineer officer must schedule all necessary MRG inspections.

You must be sure that gear sumps are cleaned after all work to the MRG has been completed. Thoroughly inspect the gear sump and remove any foreign material. Absolute cleanliness is required before filling the gear sump with clean oil.

Oil should be renovated before returning it to the sump. If new oil is used, you must be careful to remove any water or foreign particles. Cloth bags placed in the lube oil (LO) strainer baskets after major repairs is another required precaution. The bags stop foreign particles from passing through the gear train and bearings. Once the proper temperatures and pressures have been reached in the LO system, inspect the MRG for leaks and all sight-flow indicators for proper flow.

The following paragraphs discuss some of the inspections, tests, and problems that may occur to MRGs.

BACKLASH.— Backlash is the play between the surfaces of the teeth in mesh measured at the pitch circle. The backlash will increase with wear, and can increase considerably without causing trouble. On some gears that are recut, the backlash does not affect operation or

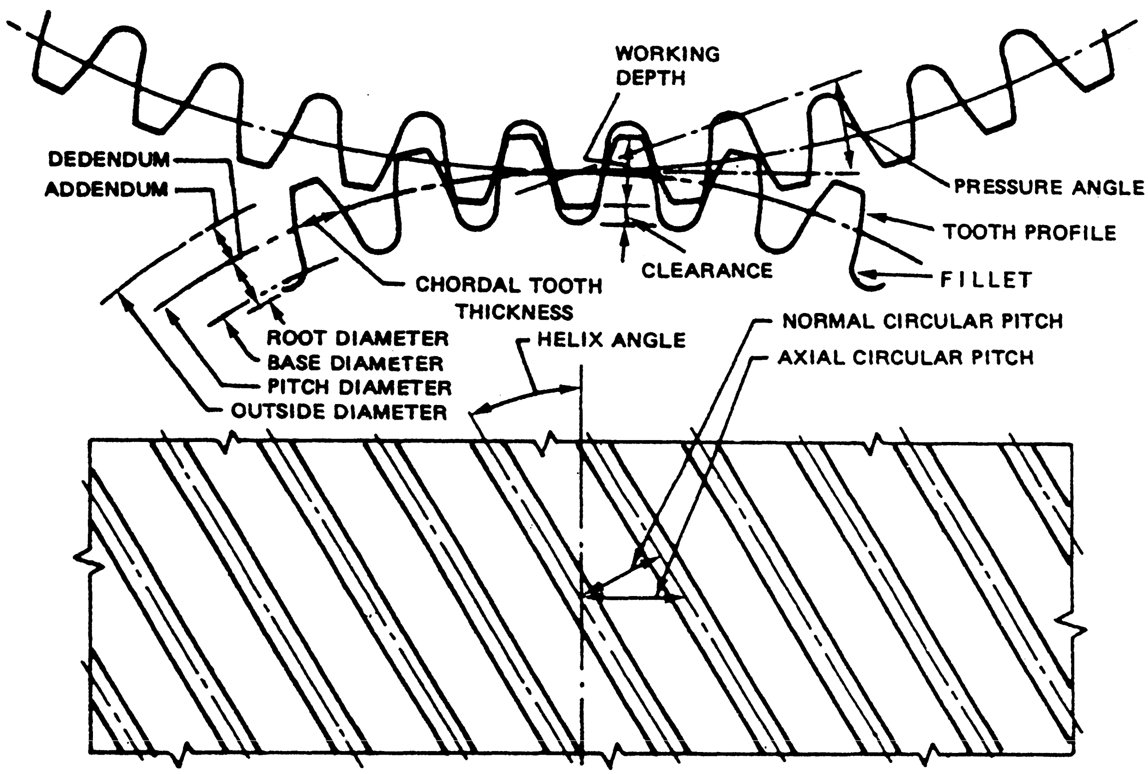

Figure 3-1.—Gear tooth nomenclature.

cause noise during ahead operation. However, a small increase in noise during astern and low-power operation may be apparent. Lack of backlash may cause noise, overloading, overheating, or failure of the gears and bearings.

DAMAGE BY FOREIGN PARTICLES.— In cases where both the pinion and gear teeth have been indented by foreign material, both should be relieved of all raised metal around the indentation. If a tooth has been dented or a foreign particle has been caught in the mesh, it will track on the mating teeth when the MRG is operated. You can hear a damaged tooth when the MRG is spin-tested. The frequency of the noise in hertz (cycles per second) will indicate which rotating element in the gear train has the damaged tooth. In double-reduction, locked-train MRGs, the damaged tooth may be on one of the four second-reduction pinions and/or one of the four first-reduction gears. The frequency will help you eliminate one of the four sets. If the damage is small, you must examine all four rotating elements until the damaged tooth is found.

FINDING DAMAGED TEETH.— Sufficient damage may be done so that just a careful visual inspection can locate the damage. If the damage is small, it may be faster to find the damaged tooth by painting the pinion teeth with a thin coat of metal-marking compound, such as prussian blue. After rotating the gears with the turning gear motor, the high spots will be shining through the coating of prussian blue.

REPAIR OF DAMAGED TEETH.— When very small foreign particles get in the gear train, they can scratch the teeth. Large particles can bend, dent, or crack the teeth. One bent or dented tooth will track on all teeth-in mesh with it. These bent and/or dented teeth can be repaired by stoning, filing, or scraping. The abraded portions of the teeth should be dressed enough to prevent the cutting of the meshing teeth. Dressing includes such actions as removal of a wire edge that is large enough to break off and pass through the mesh, and/or removal of high spots. Gear teeth should not be touched with hand tools except in an emergency! Even during an emergency, only steel scrapers or a fine file should be used, and every precaution must be taken to remove all filings or abrasive material. You should NEVER attempt to remove deep pitting or galling.

TOOTH ROOT CLEARANCE.— The designed root clearance of gears operating on their designed centers can be found in the manufacturer's technical manual drawings. You can determine the actual clearance with the insertion of a long feeler gauge, a wedge, or by the use of leads. The actual clearance should be within a few thousandths of an inch of the designed clearance and should be about the same at each end of the gear. If the root clearance is materially different at the two ends, the pinion and gear shafts may not be parallel. A difference of a few thousandths of an inch can be accounted for by errors in observation and by slight errors in machining. The amount of clearance may change a limited amount one way or another. This change is acceptable provided there is sufficient backlash so the teeth are not meshed so closely as to cause tooth interference.

GEAR TOOTH CONTACT.— Gears in mesh that are rotating in parallel and have uniform tooth contact will operate satisfactorily. Active pitting, tooth breakage, and uneven tooth contact indicate that some corrective action is required.

Satisfactory tooth contact is defined as at least 80 percent of the axial length of the working face of each tooth is in contact, distributed over nearly 100 percent of the face width. You can determine gear tooth contact using one of the following two methods:

1. Static check—Apply a thin coating of prussian blue to the pinion teeth and roll the gears with the turning gear. The compound will transfer to the gear teeth.

NOTE

Some gears are cut with a very slight taper of the teeth (helix angle deviation) to offset the effects of torsion. In such gears, full contact across the face will not be obtained by static testing.

2. Operation-Use blue or red DYKEM or copper sulphate to determine tooth contact under operating conditions. Use DYKEM for dock trials, as it will show marking with light loads. Copper sulphate shows marking after much longer and higher power operating conditions than that required for DYKEM.

TOOTH WEAR AND FAILURE.— Wear is defined as the removal of metal from the gear teeth. Normal wear is the removal of metal at a rate that does not impair the satisfactory operation of the gear. If proper tooth contact is obtained when the gears are installed, little trouble should be encountered in respect to wear. Excessive wear cannot take place without metallic contact. Proper clearances, inspections for removal of high spots, and/or adequate supplies of lubricating oil can prevent excessive wear. If the lubricating oil supply should fail and the teeth become scored, the gears must be overhauled at the first opportunity.

Pitting, particularly along the pitch line, may occur in the first few months of service. This pitting (often referred to as connective pitting) usually stops after a short time, and no further trouble is experienced. Corrective pitting requires only one precaution. You must be sure that no flakes of metal are allowed to remain in the LO system. Remember, very minor pitting does not affect operation. Pitting in new gears is due to very slight high areas. These high areas are removed by the pitting. This condition is corrective and will stop. However, pitting that continues can result in progressive deterioration of the gear (fig. 3-2).

Scoring is characterized by transfer of metal from one sliding surface to another. Scoring in gear teeth is caused by contact of the tooth tips due to insufficient tip relief or lack of lubrication.

Dirt tracks are caused by foreign particles passing through the mesh. The gear teeth are marked in the same location on each meshing tooth. Prominent high spots caused by foreign particles require removal. Removal of foreign particles avoids problems such as load concentration, pitting, or tooth breakage.

Wire edge caused by plastic flow of metal results in a “fin” at the outside diameter of the tooth. If the fin is heavy, it must be removed. If not removed, it may break off and pass through the mesh.

Cracked teeth are normally caused by fatigue, but may be caused by shock. Cracked teeth like those shown in figure 3-3 will break if operation of the MRG is continued. The cracks are clearly shown by indicating dyes used for inspection.

Tooth fatigue breakage is caused by repetitive cycling at a load greater than the fatigue strength of the material. Tooth fatigue is progressive. A short crack appears first, and then propagates. Characteristic “oyster shell” lines can usually be seen. Figure 3-4 shows a typical broken tooth.

Alignment

The gear train is in alignment when the gear and the pinion are parallel. That is, the axis of the two shafts are in the same plane and equal distance from each other at

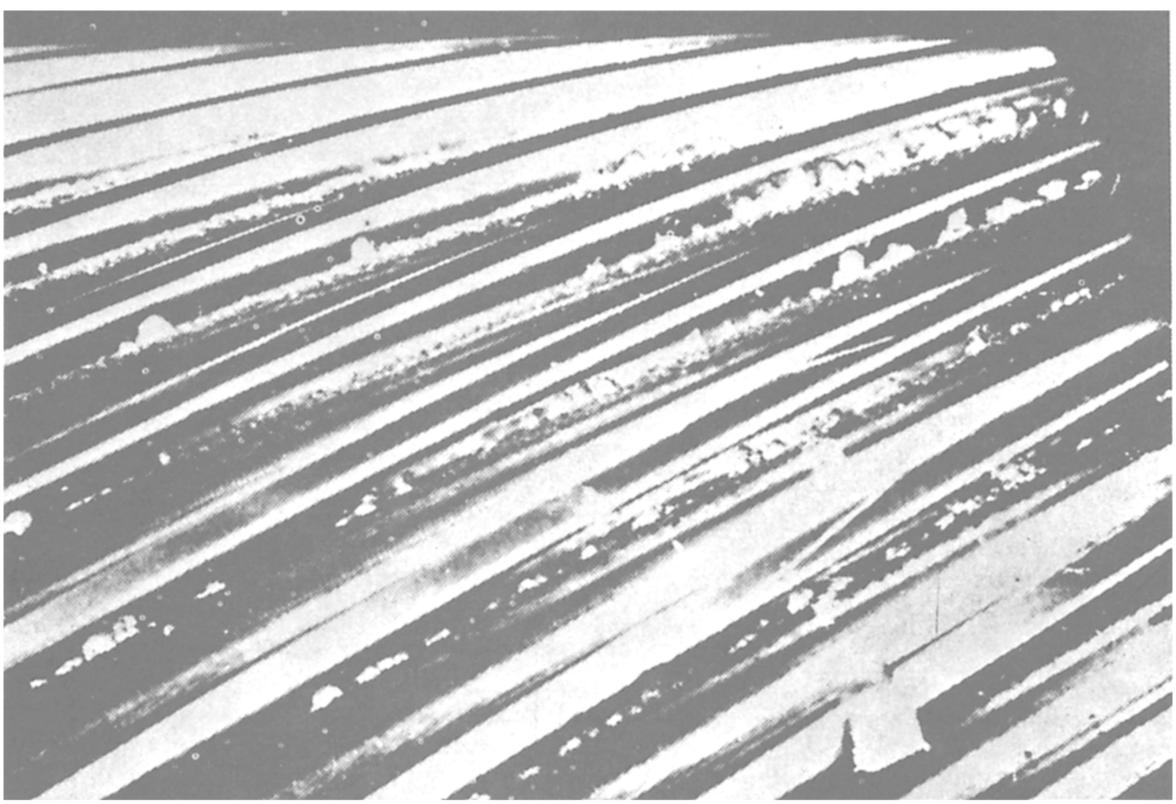

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Figure 3-2.—Progressive destructive pitting.

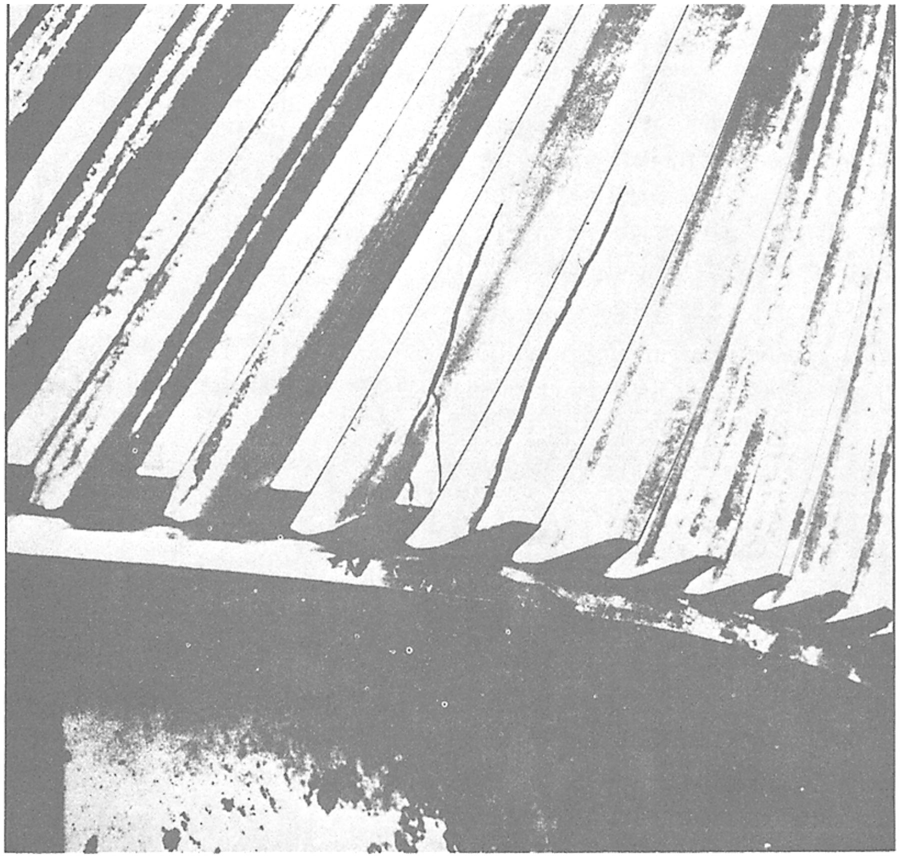

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Figure 3-3.—Cracked teeth

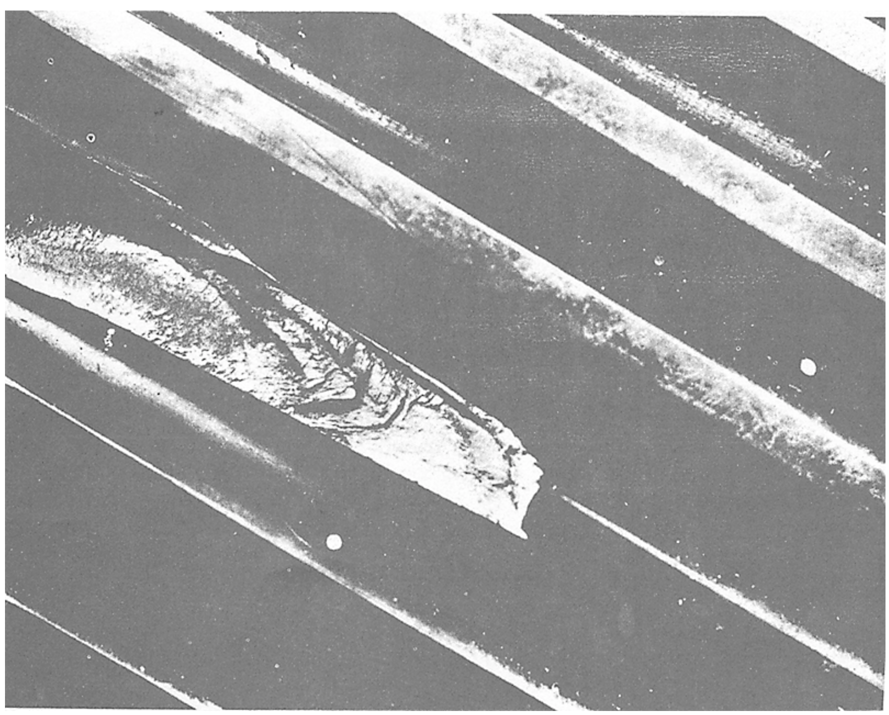

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Figure 3-4.—Broken tooth.

all points. In service, the best indication of good alignment is good tooth contact.

The technical manual furnished with each gear installation describes the procedures for determining the proper depth of mesh and parallelism of gear and pinion shafts. The length of tooth contact across the face of the gear teeth is the key to satisfactory alignment of reduction gears.

Poor alignment between the line shaft and the MRG may be detected at the reduction gear. Uneven loading of the low-speed gear train and noisy operation in certain speed ranges are two common results of poor line shaft to MRG alignment.

The most favorable alignment position of the main engine to the reduction gear is when they are concentric at full power at the proper operating temperature. The flexible high-speed coupling is designed to handle the transient condition of slight misalignments as the machinery comes up to temperature. The two most common forms of misalignment between the prime mover and the driven shafts are angular and parallel offset, as shown in figure 3-5.

The object of the alignment is to locate the turbine so the axis of the spindle will be concentric with and parallel to the axis of the reduction gear input pinion shaft. Attaining alignment is complicated by the fact that the turbine, reduction gear, and foundations all expand as they are heated during operation to the hot running conition. Another factor is when operating pinion shafts move higher in their bearings under the influence of the hydrodynamic oil film and tooth pressure. These changes in position have been predetermined by the manufacturer, and you can find the offset readings in the appropriate technical manual for the installation.

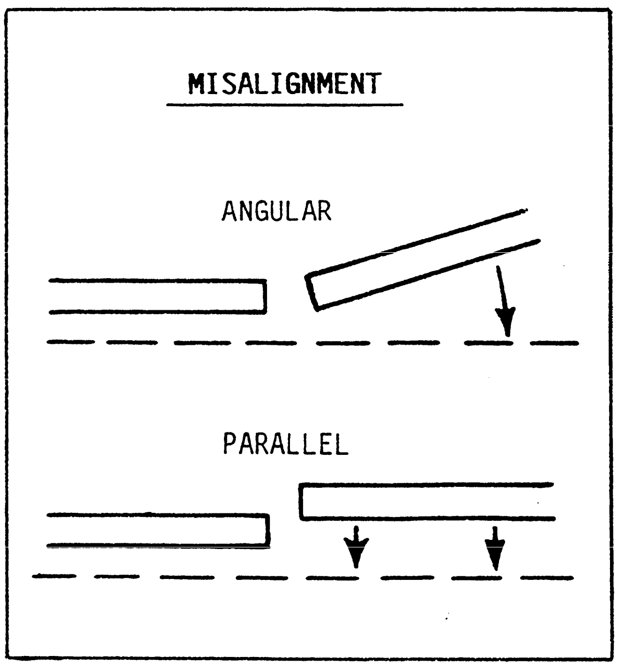

Figure 3-5.—Angular and parallel misalignment.

MAIN THRUST BEARING CLEARANCE MEASUREMENTS

As you have already learned in *Gas Turbine Systems Technician (Electrical) 3/Gas Turbine Systems Technician (Mechanical) 3,* volume 1, NAVEDTRA 10563, propeller thrust is transferred from each propulsion shaft to the hull through a Kingsbury main thrust bearing (fig. 3-6). The Kingsbury thrust bearing uses the wedge-shaped oil film lubrication principle. This principle is based on an oil film between two sliding surfaces tends to assume a tapered depth with the thicker film at the entering side. In a Kingsbury assembly, eight bearing shoes are installed on each side of the thrust collar. Therefore, eight separate wedge-shaped oil films are installed on each thrust face. Since the bearing shoes are free to tilt slightly, the oil automatically assumes the taper required by shaft speed, loading, and oil viscosity.

The main thrust bearing assembly consists of the bearing housing, two thrust rings, and a thrust collar. The housing, thrust rings, and thrust collar facings are all split horizontally. Each thrust ring is made up of 8 steel thrust shoes with tin babbitt facings, 16 leveling plates, and a retainer ring. The thrust collar has a two-piece removable steel thrust face attached to each side. Each thrust shoe contains a hardened shoe support with a spherical face. The support bears on the upper leveling plate and the spherical face allow the thrust shoe to pivot or tilt slightly in all directions. This arrangement allows the bearing to operate on the free-wedge film lubrication principle. One thrust shoe on each side is fitted with a resistance temperature element (RTE).

Due to the spring isolation system, main thrust bearing clearance measurements are no longer taken with a depth micrometer. All measurements are now taken with a dial indicator that measures the deflection of the propulsion shaft at the main flange. There are two methods (static and dynamic) used to create shaft deflection. The method used depends on the ship class. The static method must be used on CG-66 and above and all DDG-51 class ships. The dynamic method is

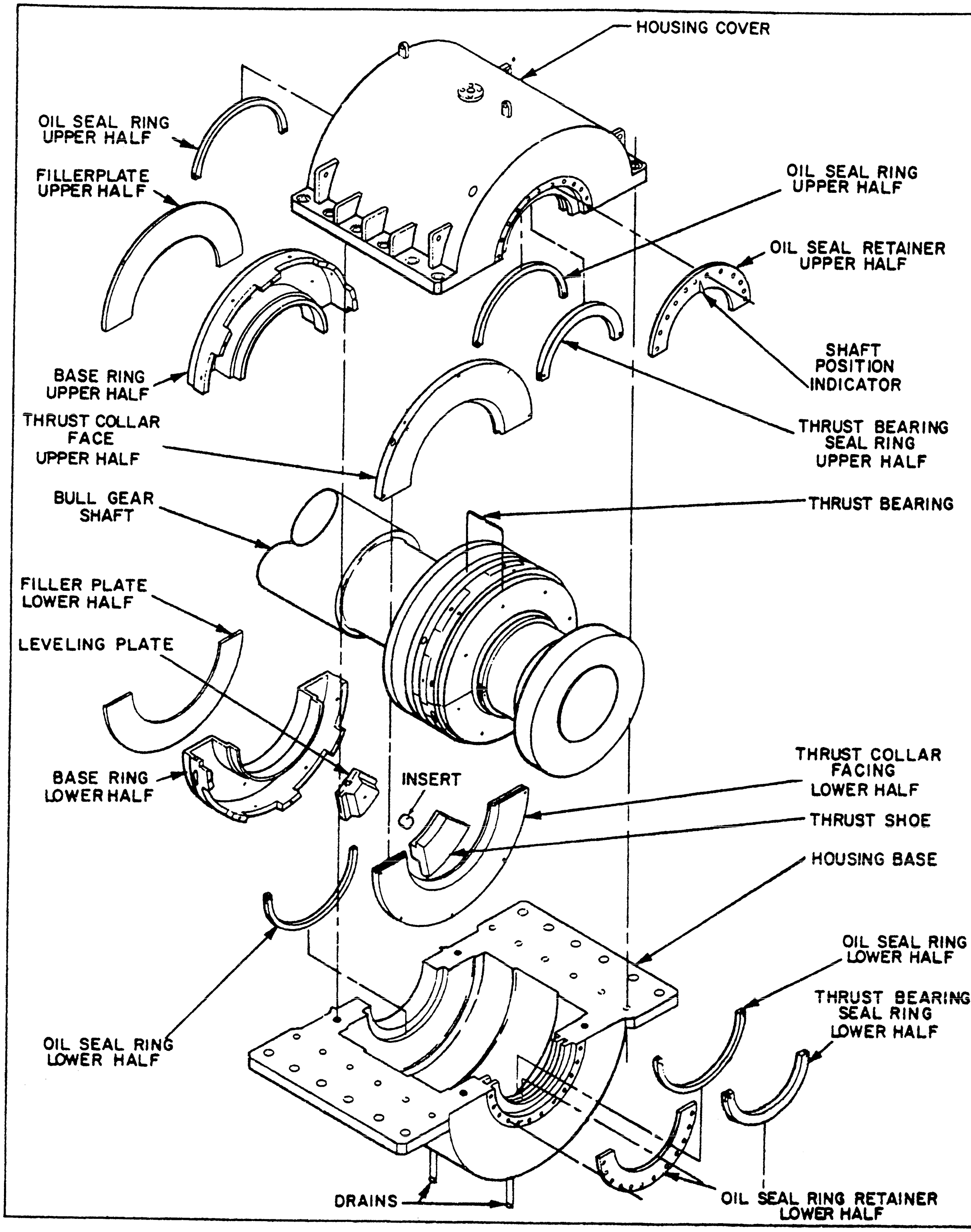

Figure 3-6.—Thrust bearing and housing.

used for all other gas turbine-powered ships. Take a brief look at these two methods to measure main thrust bearing clearances.

Static Method

In the static method, a dial indicator is mounted on top of the aft end of the thrust housing with the zero pointer against the forward face of the first shaft flange aft of the thrust bearing. While the controllable and reversible pitch (CRP) system is operating, the propeller pitch is advanced to 100 percent ahead with the local controls at the oil distribution (OD) box. The electric CRP pump is then secured, and the hub servo bottomed-out by use of the emergency pitch hand pump. The reduction gear is then rocked with the turning gear motor by use of the ratchet wrench to turn the motor. This movement assures that the shaft is bottomed-out on the ahead thrust collar.

If the clearance is within normal limits, and no abnormal conditions exist, log the readings and put the system back in service. If the clearance is not within normal limits, or a noticeable increase or decrease is measured, inspect the thrust bearing. Check for any abnormal rendition (scored or wiped shoes) and take corrective action as necessary.

NOTE

Approximately six to eight turns of the wrench in either direction will remove the backlash; turn an additional two to four turns. Repeat once or twice in each direction with 2500 to 3000 psi on the emergency hand pump.

Dynamic Method

Like the static method, the dynamic method requires a dial indicator and instrument placement to measure the shaft deflection. Dynamic measurements are done while the propulsion shaft is in operation. Communications must be established from the area of the thrust bearing to the station with throttle control. The ship must be operated in the ahead direction at a 1/3 bell for 10 minutes. Allow the shaft to coast to a stop and then position the dial indicator against the shaft flange. After the indicator is in place, operate the ship in the astern direction at a 1/3 bell for 5 minutes. After allowing the ship to coast to a stop, record your deflection reading. This procedure is repeated two more times (three times total). You must use the average of these three readings to obtain the main thrust bearing clearance. Designed thrust bearing clearance is 0.030" to 0.045", with a maximum of 0.080".

For specific detailed information on these procedures, consult the applicable PMS maintenance requirement card (MRC) or manufacturer's technical manual for the speed decreaser gear installed on your ship.

CLUTCH AND BRAKE ASSEMBLIES

Depending on the type of ship to which you will be assigned, you will encounter either one or both of the two types of current clutch assemblies used on gas turbine-powered ships. The first and most widely used clutch assembly is the synchro self-shifting (SSS) type. This type of clutch assembly is installed on all CG-47, DDG-51, and FFG-7 class ships. The other type is a pneumatically operated, forced-synchronization type of clutch assembly. The forced-synchronization clutch assembly is installed on DD-963 and DDG-993 class ships.

Along with the two types of clutch assemblies, there are two types of power turbine (PT) brake assemblies installed on gas turbine-powered ships. The type of brake assembly used depends not only on the ship class, but also on the type of clutch assembly installed.

In this section, we will briefly discuss the normal operation and maintenance related to both types of clutch assemblies and all the brake assemblies. Because of the complexity, the elaborate control system, and the large number of labor hours required for maintenance, the Navy is gradually phasing out the forced-synchronization type of clutch. Because of this phase-out, we will focus our discussion on the maintenance practices associated with the SSS type of clutch.

NORMAL OPERATIONS

Both types of clutch assemblies perform the same function. They connect a GTM or the GTMs to the MRG to drive the propulsion shaft. It is not the function, but the method of clutch engagement that varies drastically between the SSS and forced-synchronization clutches.

Forced-Synchronization Clutch

The forced-synchronization type of clutch requires ship's service air and MRG LO availability before normal engagement can occur. The clutch is made up of a friction pack and dental clutch assembly. The friction pack is needed to bring the GTM input shaft speed to within 11 rpm of the first reduction pinion. Once the speed permissive are met, air pressure is applied to the dental clutch to complete engagement. When the dental clutch is engaged, all torque is transmitted from the GTM input shaft to the MRG's first reduction pinion. This clutch assembly also houses a friction-type PT brake that serves two purposes. The primary purpose of the PT brake is to stop and hold the PT stationary. If the PT brake is used with an engaged clutch, the PT brake also acts as a shaft brake. The forced-synchronization clutch is being phased out of Navy service in favor of the SSS clutch.

Synchro Self-Shifting Clutch

Like the forced-synchronization clutch, the SSS clutch performs the same functions by transmitting engine torque through the input shaft to the MRG first reduction pinion. It does not, however, require any external controls to perform the engagement sequence. For clutch engagement to occur, the SSS clutch requires only that the input shaft speed be greater than the speed of the first reduction pinion. The SSS clutch is fully automatic. By design, centrifugal force causes the main sliding member to move and engage with the output assembly.

Depending on the ship class, the SSS clutch system uses two different types of PT brake assemblies. The CG-47 and DD-963 class ships have a similar PT brake assembly. The PT brake assembly is an internally housed friction clutch design that is mounted to, but operates independently of, the SSS clutch assembly. There is one main difference between the CG-47 class ships brake and the one installed on the DD-963 class ships. The CG-47 brake cannot be used as a shaft brake. Because of the SSS clutch design, even if the brake is applied with the clutch engaged, the clutch will disengage once the PT input speed drops below the speed of the first pinion.

The other type of PT brake assembly is the one installed on the DDG-51 and FFG-7 class ships. This is a single-disc caliper brake assembly that is externally mounted to each PT input shaft. These brakes are used with the SSS type of clutch in which their only purpose is to stop and hold the PT stationary when required.

The last type of brake assembly we will discuss is the shaft brake assembly. Shaft brake assemblies are installed only on FFG-7 class ships. The shaft brake assembly is also a single-disc caliper brake assembly, such as the PT brake assembly we just described Consisting of two complete units, a shaft brake assembly is mounted on each starboard first reduction quill shaft. Once all permissive are met, the single purpose of this brake is to stop MRG rotation.

MAINTENANCE OF EQUIPMENT AND COMPONENTS

The maintenance of the clutch and brake assemblies and associated equipment and components is normally done according to the PMS. General cleaning, tests, and inspections will be your primary concern. Because of the good operational track record associated with the SSS clutches, troubleshooting and repairs should be minimal. This good operational record is the main reason the Navy is phasing out the forced-synchronization type of clutch in favor of the SSS clutch.

In the following paragraphs, we will discuss some general maintenance and repair practices associated with clutch and brake assemblies. You, the GS supervisor, must be familiar with these practices so you can properly supervise maintenance and repairs.

Cleaning

The cleaning of the clutch and brake assemblies is done primarily when the MRG is cleaned Cleaning the clutch assembly is limited to external cleaning. Pay attention to the areas around the inspection and access cover plates and the clutch position indicating ports.

The cleaning of externally mounted brake assemblies requires a little more effort. To properly and thoroughly clean any of the external brake assemblies, you must remove the guard screen. Once you remove the screen, pay particular attention to removing any dust accumulation on the brake and to the cleanliness of the disc. It is important that you keep dust accumulation to a minimum. Once dust mixes with oil, it can be deposited on the brake disc or absorbed into the pads. Excessive dust and/or oil accumulation can seriously degrade the brake's operation.

Tests and Inspections

Unless a casualty occurs to either the clutch or brake assemblies, all tests and inspections are performed according to the PMS. There are no tests or inspections related to SSS clutch assemblies, unless you are assigned to a CG-47 class ship. Remember, that particular type of SSS clutch has an internal PT brake assembly. The ship's maintenance action plan periodically requires that an inspection of the disc assembly be made and the clearances between the discs measured.

Additionally, you must check the externally mounted PT and shaft brake assemblies on a regular basis according to the PMS. These inspections normally include checking the brake pad thickness measurements, rotor condition, proper operation of air or hydraulic actuators, and proper lubrication of vital moving parts.

TROUBLESHOOTING

Because SSS clutches are reliable, problems that require troubleshooting are usually minimal. Like all other gear-driven assemblies, SSS clutches have a tendency to wear and produce noise with age. Normal failures are usually limited to faulty position indicator switches and failures related to the PT brake assemblies. We will not dwell on the clutch assemblies, but move on to some of the problems related to the installed brake assemblies and the ways in which you, the GS supervisor, can better identify them.

The basic operation of both the PT and shaft brakes is the same as the disc brake system installed in most automobiles. All brake systems require some type of medium (air, oil, or air and oil) to force the caliper piston against the brake pad which, in turn, is pushed against the disc. This action slows the rotation of the disc until the disc stops. Next are some common malfunctions that may occur in this system and ways that you can isolate the cause.

Failure to Engage

There are several problems that can cause a brake to fail to engage. You must understand the operating principles associated with the system. First, check to see if there is sufficient air or oil pressure for operation. It is pretty obvious that if the activating medium (air or oil) is missing, this condition should produce an alarm at the console.

Once you determine that the activating medium is available, you should try the manual control. If the manual control works, you should consider an electrical fault as the problem source. If the manual control does not work, you should continue troubleshooting. If the pressure regulator is not working, the supply cutout valve (if installed) may be closed, or there may be a blockage or leak in the supply line. These are all possible causes for the failure. The last possibility to check is the electrical control. Did the brake actually engage? If the brake engages, but you do not receive a brake engaged indication, just look at the PT speed to verify a slowing down or a stop. If the PT has stopped, your indicator light may be out or the indicator switch may be bad. If the PT does not stop, you may need help locating where the command signal is lost.

Failure to Release

When a brake fails to release, the three most common causes are a command problem, a bad position indicator switch, or a bad indicator light. If none of these are the cause, you should check for a binding caliper and weak or damaged return springs.

Failure to Stop Rotation

When the brake applies but does not stop rotation, the most common causes are insufficient actuating pressure, contaminated brake pads, a damaged rotor (disc), or a binding caliper piston.

ALIGNMENTS AND ADJUSTMENTS

Basically, the only components that have any adjustments or alignment checks are the PT and shaft brake systems. Normally, all of these adjustments or alignments are performed as requirements resulting from a PMS inspection.

REMOVAL AND REPLACEMENT OF COMPONENTS

The removal and replacement of a clutch maybe performed by your ship's personnel if there is sufficient time or if a casualty occurs. Most of the time, however, the engineer officer will opt to have an outside activity perform the work.

On the other hand, the brakes and their subsystems can be easily maintained by your ship's maintenance technicians and personnel.

LINE SHAFT (SPRING) BEARINGS

The line shaft (spring) bearings are self-aligning, oil-lubricated journal bearings. Each bearing is a self-contained assembly with its own oil reservoir that contains 2190 TEP oil.

An oil disc (ring) clamped to the shaft is used in each bearing to deliver oil to the upper bearing and journal surfaces. As the disc rotates, it picks up oil from the bearing reservoir and carries it to the oil scraper on the upper shell. The scraper removes oil from the disc and directs it to the upper bearing lining. A clear sight cover on the bearing housing allows visual confirmation of the oil disc operation. Figure 3-7 shows a typical disc-oiled line shaft bearing.

All bearing pedestals have an oil level rod and an oil reservoir thermometer for checking oil level and temperature. A resistance temperature detector (RTD) is installed in the lower bearing shell of each oil-lubricated bearing. The RTDs provide for remote readouts of each bearing's temperature on the digital demand displays.

Clearances are taken with a depth micrometer through a port in the upper bearing housing, which contains the anti-rotation pin. The original installation readings are stamped into the flat surface adjacent to this port. You must take readings according to PMS requirements or when an abnormal condition exists.

For information on maximum wear limits and repair procedures, you should consult the appropriate manufacturer's technical manual and *NSTM,* chapter 244, "Shafting, Bearings, and Seals."

STRUT AND STERN TUBE BEARINGS

Each propeller shaft extending aft of the stern tube is supported by two struts, each containing a seawater-cooled bearing. Figure 3-8 shows a typical strut bearing.

Stern tube bearings are in constant contact with the seawater surrounding the stern tubes. The clean seawater that passes through the stem tube seals from the ship's seawater service system or firemain system (in emergencies) also flows through the stern tube bearings. Stern tube bearings are identical to forward strut bearings. However, aft strut bearings are roughly 5 inches larger in diameter and twice as long as stem tube bearings. Remember that stem tube bearings are not remotely monitored.

PROPULSION SYSTEMS

The ship's propulsion thrust is provided by hydraulically actuated propellers. In the Gas *Turbine Systems Technician (Electrical) 3/Gas Turbine Systems Technician (Mechanical) 3,* volume 1, NAVEDTRA 10563, you were provided with a complete description of propulsion systems and how they operate. As a gas turbine supervisor you need to be knowledgeable and experienced with a variety of gas turbine propulsion

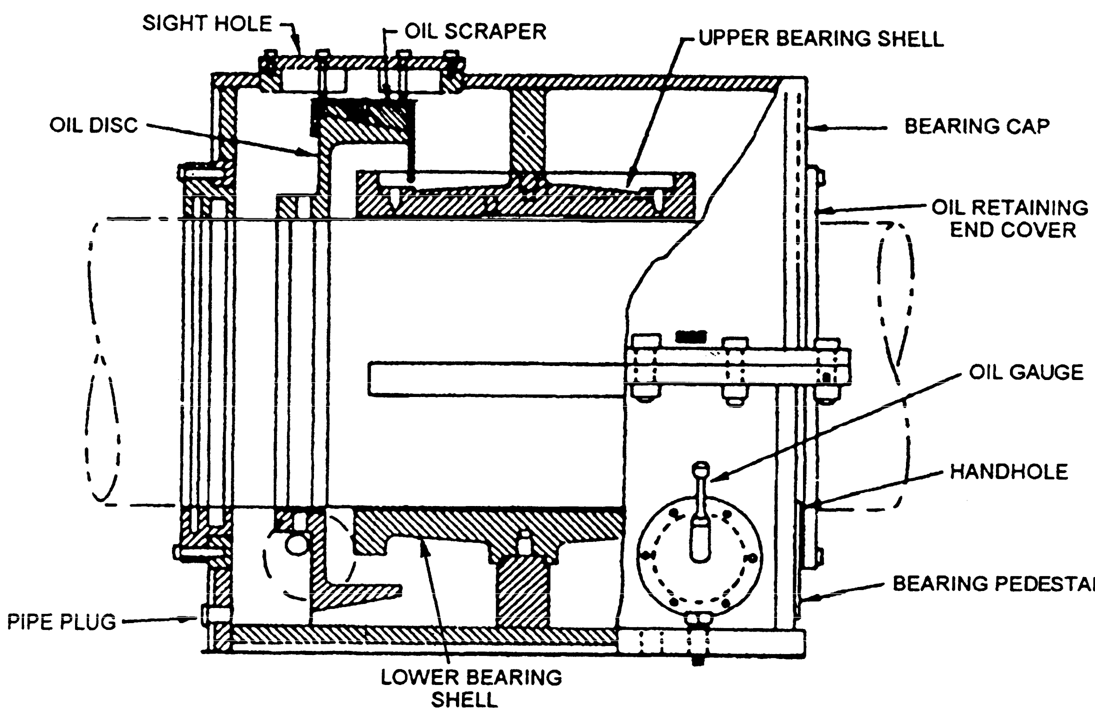

Figure 3-7.—Disc-oiled line shaft bearing,

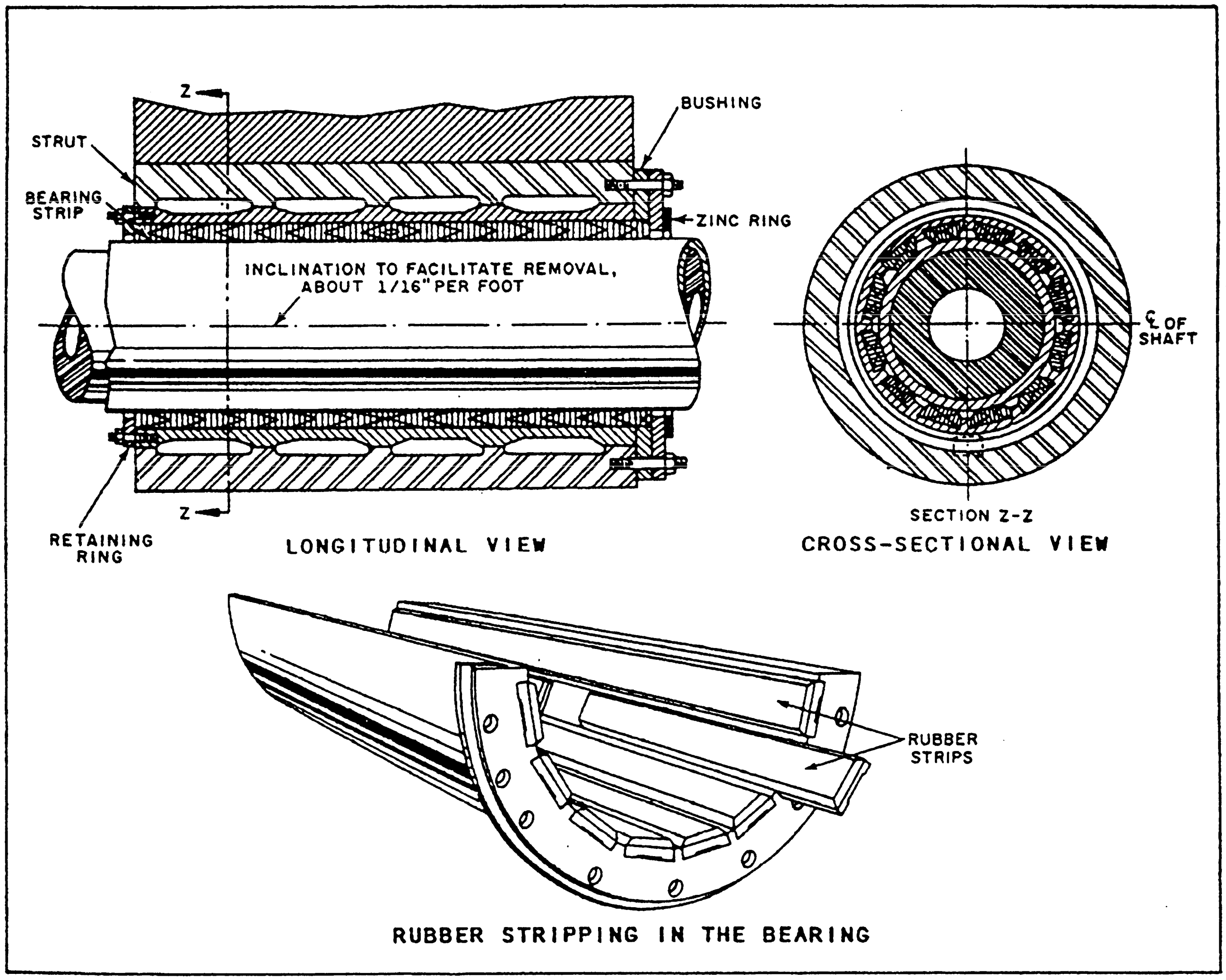

Figure 3—8.—Details of a typical underwater strut bearing.

system operations. This proficiency combined with leadership skills will assist you daily in your supervisory role.

CRP/CPP SYSTEMS OPERATIONS

In this section, we will briefly cover routine and emergency operations of the CRP/CPP systems. Since the various ship systems are functionally the same, we will not indicate ship type except when necessary. However, if you need to review the basic design and operational characteristics of these systems, consult *GSE3/GSM3,* volumes 1 and 2.

Routine Operation

Propeller pitch changes can be made through the full range of travel in one continuous movement. The pitch change can be made in automatic, remote manual, or local manual operating mode. The maximum rate of pitch change, 30 seconds from full ahead to full astern, is determined by the response of the hydraulic system. Pitch change rate is NOT determined by the speed of movement of the control lever.

The CRP/CPP systems were designed to maximize the ability of the ship's propulsion GTE to accelerate and decelerate rapidly and to enhance maneuverability. To further enhance the systems' performance and to make them more "user friendly," some subtle changes were made. These changes have been installed on the newest gas turbine ship platform, the DDG-51 class ship. The following sections will explain some of these changes and the contributions they make to system performance.

As a GS supervisor, you should be aware of the changes incorporated in the CRP/CPP systems installed on the DDG-51 class ships and the advantages they offer in the performance and operation of these systems. Look at some of the components that have changed in the OD box and the pitch indicating system.

OIL DISTRIBUTION BOX.— On DD-51 class ships, the configuration of the OD box has undergone several changes. The emergency pitch pump hose connections have been moved from the bottom of the OD box to the low-pressure chamber cover, as shown in figure 3-9. This change makes it easier for the operator to make the connections during testing or in case of an emergency.

PITCH INDICATING SYSTEM.— A significant change was made to the pitch indicating system. There are now two types of pitch indicating systems installed on DDG-51 class ships. One system is temperature compensated while the other is electronic. Because these two systems operate independently, DDG-51 class ships have both a normal and alternate means of measuring propeller pitch.

Temperature-Compensated Pitch Indicator System.— On DDG-51 class ships, the temperature-compensated pitch scale platform is rigidly

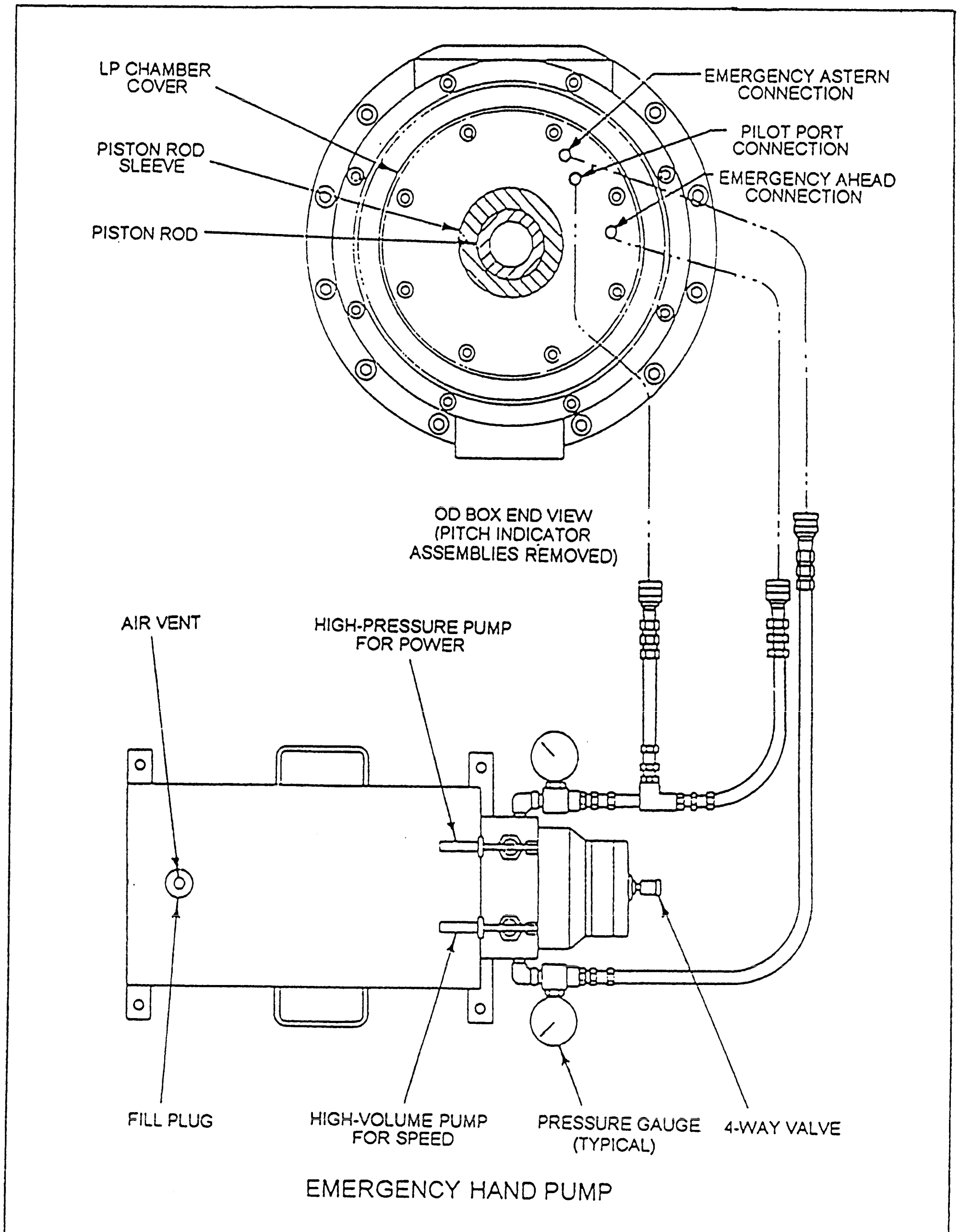

Figure 3-9.—Emergency pitch arrangement.

connected to the OD box. The scale platform holds the feedback potentiometer and local pitch indicator, as shown in views A and B of figure 3-10. This indicator is defined as temperature compensated because the indicator arm is connected to the prairie air tube. Look at view B. The prairie air tube is normally pressurized with air at a controlled temperature, and will have a fixed amount of thermal growth. Hence, the indicator arm is provided with a thermally stable (temperature-compensated) surface, a primary means to sense and indicate propeller pitch, and a means to relay that information to the machinery control system (MCS).

Electronic Pitch Indicator System.— In addition to the temperature-compensated pitch indicator installed on the OD box of DDG-51 class ships, an electronic pitch position transducer is installed behind a

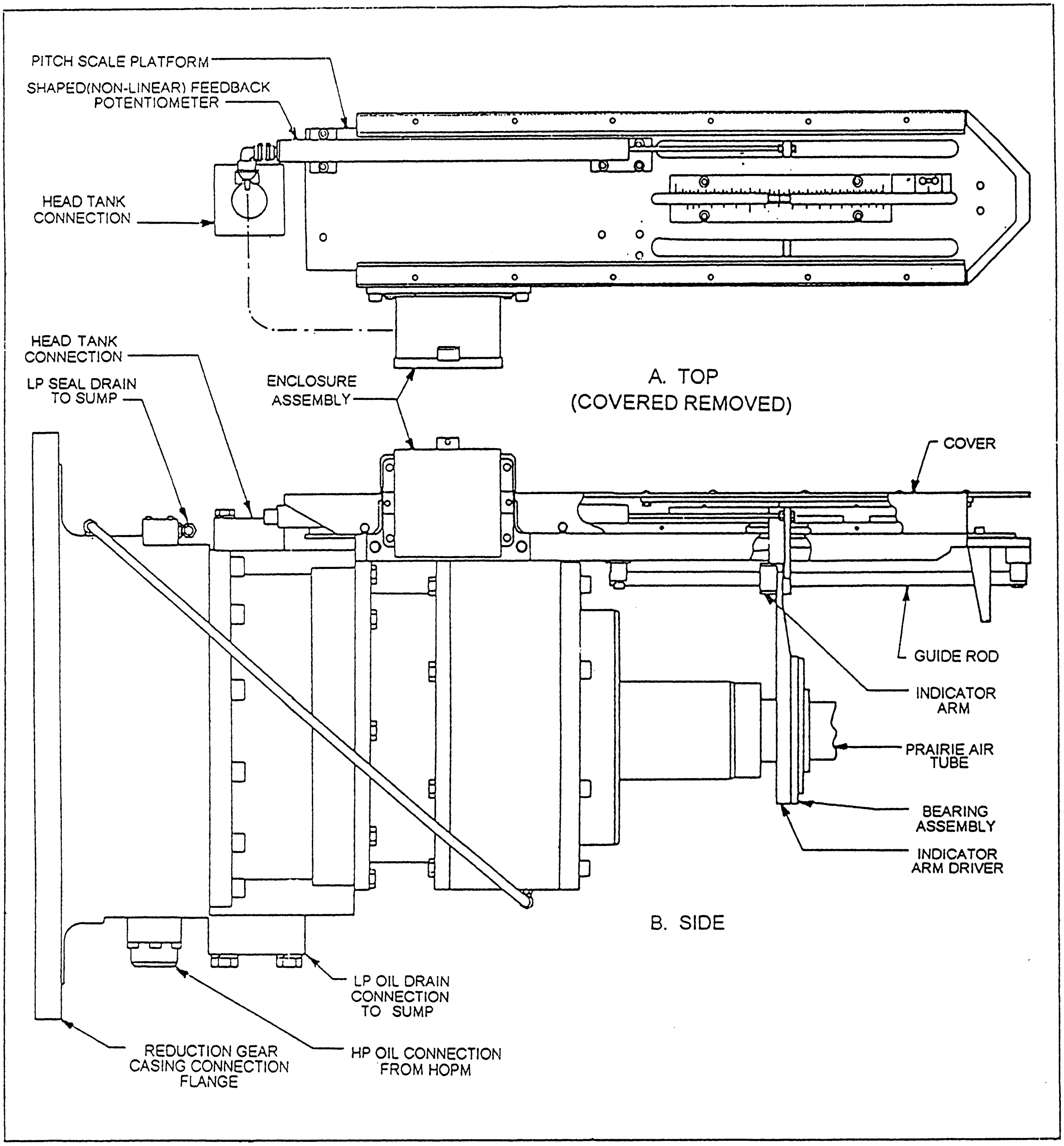

Figure 3-10.—Pitch scale platform showing temperature-compensated pitch indicating system.

cover plate on the propeller hub. This arrangement is shown in view A of figure 3-11. The electronic pitch indicator receives an input from a sensor assembly mounted inside the propeller hub, as shown in detail in view B of figure 3-11. The sensor (wand) extends from the small electronics package (handle) located in the hub cone and cover into an axial hole drilled into the piston rod. The hole contains a magnetic ring that allows the sensor to measure propeller pitch position.

The electronic pitch indicator system also contains a stationary electronics cabinet, rotary transformer, and rotating electronics cylinder. The stationary electronics cabinet is mounted adjacent to the OD box. It contains the circuitry to provide a 10-kHz excitation signal to, and receive a propeller pitch position feedback signal from, the rotary transformer. The cabinet also has two light emitting diode (LED) displays that show propeller pitch in both feet and percent of design ahead and astern pitch. The rotary transformer contains both the

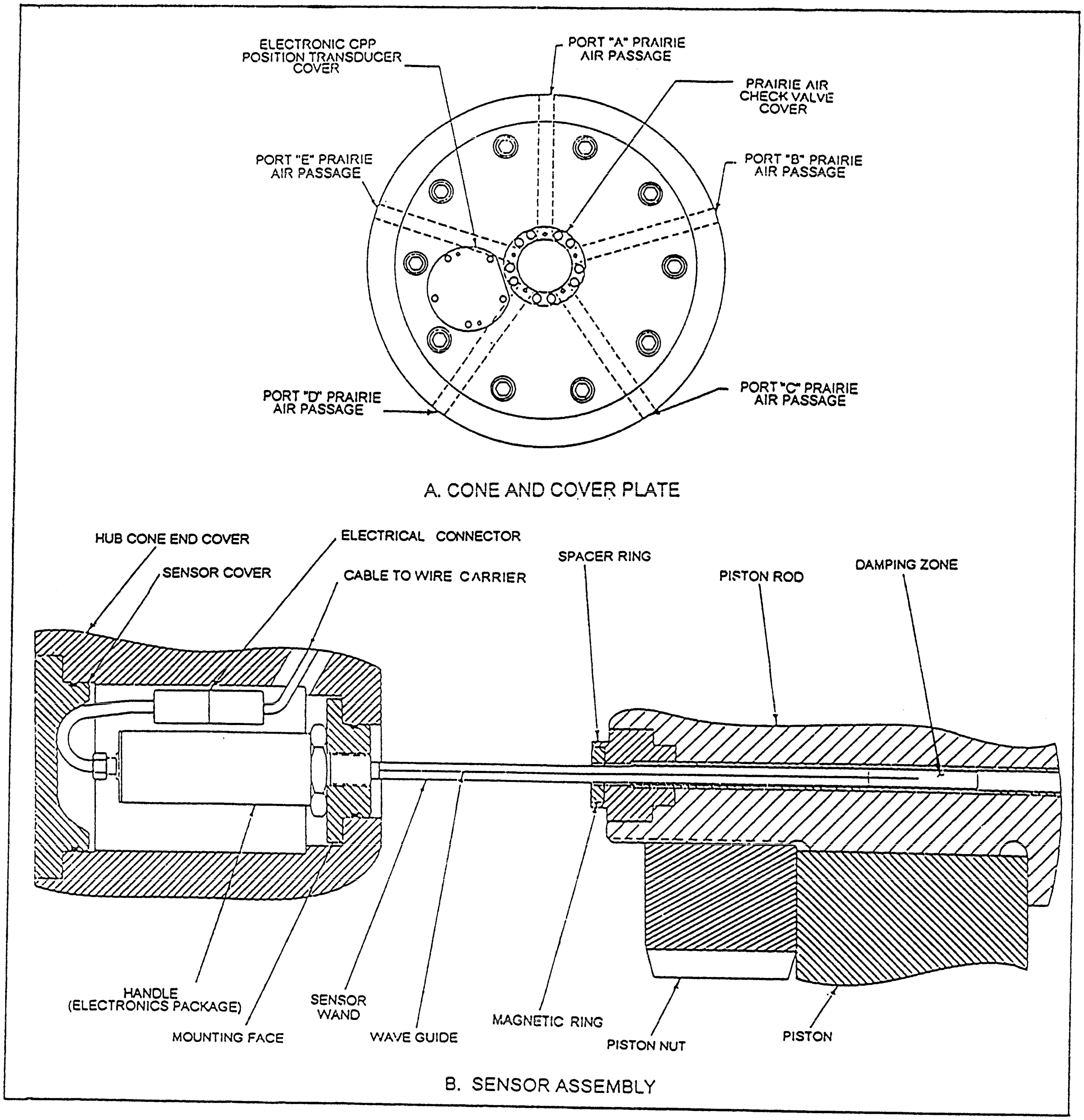

Figure 3-11.—Hub-mounted electronic pitch position transducer.

transducer excitation and output coils. The rotating electronics cylinder rectifies the rotary transformer excitation to 24 V dc for powering the hub-mounted transducer and receives the transducer output. The rotating electronics cylinder is attached to the prairie air tube extension at the end of the OD box. The cylinder electronically regulates the transducer output to 3 kHz for the rotary transformer. For a more detailed description of this system's operation, consult the DDG-51 propulsion plant manual or the DDG-51 CPP system technical manual.

Emergency Operation

If the electronic pitch control system becomes inoperative, you must instruct your personnel to shift to manual control at the OD box. Manual operation is accomplished by shifting the two changeover valve levers on the manifold block assembly from AUTO to MANUAL. The propeller will remain at the pitch set initiated at the time of the change. You can change the pitch by operating the control valve lever and observing the local pitch indicator.

Once the problem with the electronic controls has been corrected, shift back to automatic operation. Before initiating the change from MANUAL to AUTO, be sure the pitch command signal from the controlling remote station matches the actual pitch setting at the time the mode change (manual to auto) is initiated. Shift the two changeover levers simultaneously to the AUTO position. If the command signal and actual pitch settings do not match, the pitch will automatically change to the commanded pitch upon completion of the mode change.

If the local manual mode is inoperative, shift the changeover valves to the OFF position, secure the hydraulic oil power module (HOPM), and set emergency pitch according to engineering operational sequence system (EOSS) procedures.

> **NOTE**
>
> When operating in the emergency ahead mode, the OD box temperature must be monitored continuously. You may have to apply cooling water to the OD box housing to maintain the temperature below 160°F. Remember, the most effective and preferred method to maintain OD box temperatures is the use of prairie air.

CRP/CPP SYSTEM INSPECTIONS

The following is a list of routine checks and services that should be made to the CRP/CPP system while underway:

1. Hourly
   a. Check the OD box temperature, 160°F maximum (150°F DDG-51 class).
   b. Check the hydraulic system oil pressures; ensure all comply with EOSS.
   c. Check the prairie air rotoseal for leaks.
   d. Check the ΔP for the online strainers and falters.
2. Each watch
   a. Check the hydraulic oil level in the sump and head tank.
   b. Inspect the hydraulic system for leaks and vibration.
3. Before entering port or any restricted maneuvering situation
   a. Verify the control system response by decreasing pitch slightly and noting response.
   b. Verify the pump (attached or electric) operation depending on ship class.
4. When the system is secured
   a. Check the oil level in the head tank. If the head tank oil level is low, replenish oil as necessary by operating the system until the oil level returns to normal.
   b. Inspect the oil for water contamination periodically.

CRP/CPP TROUBLESHOOTING AND MAINTENANCE

Even though the CRP/CPP systems installed on a gas turbine-powered ship use the same types of pumps and fluids as the MRG LO system, they are much more sophisticated. Paying close attention to system operating pressures is your most important step in successfully troubleshooting any problem and making the appropriate repairs. The following paragraphs will describe some of the common problems associated with system pressure and the methods you can use to identify the component or components that might be causing the problems. The two most common casualties that you can identify just by noting variations in the system pressure are loss of pitch control and loss of hydraulic oil pressure.

Loss of Pitch Control

A loss of pitch control can be caused by either a mechanical or an electrical failure. Mechanical failures tend to occur more frequently. You should be aware, however, that electrical problems can occur that will occasionally produce a loss of control. Normally, an electrical problem, such as a broken or loose cannon plug or loss of feedback position, will require the system to be shifted to manual control. In DDG-51 class ships, however, the CPP system has both a normal and an alternate system by which the pitch position can be monitored. If the normal system fails to provide command or feedback data, the CPP system can be shifted to the alternate system. When investigating a loss of pitch control on any gas turbine-powered ship, you must be aware of the components in the CRP/CPP systems that are most likely to fail. The following are some of the components you will have to monitor most frequently.

ELECTROHYDRAULIC SERVO VALVE.— The most common component failure is the electrohydraulic servo valve. This valve is the primary component for remote operation and control. If this valve were not installed, all operations would require personnel to be stationed at the OD box at all times.

You can easily identify a faulty electrohydraulic servo valve. Any of the following symptoms should alert you as to the possible failure of this valve:

- Pitch fails to respond to a desired integrated throttle control (ITC) change.
- Pitch changes (fluctuations) occur without a pitch change command.
- Hub servo pressure increases steadily without a change in system demands.

AUXILIARY RELIEF VALVE.— A faulty auxiliary relief valve also will cause a loss of pitch control. If the valve fails in the open position, all of the control oil will be ported back to the sump. Pitch cannot be changed without control oil to position the auxiliary servo piston in the OD box. In addition to the loss of pitch control, you should investigate any loss of pressure. You should be able to spot a pressure loss by checking the HOPM pressure gauges. This should be one of your first steps in checking the system.

REDUCING VALVE.— A faulty reducing valve is another cause for a loss of pitch control. If the reducing valve fails in the closed position, the flow of control oil will be cut off to the auxiliary servo piston, and pitch will fail to respond. Like the auxiliary relief valve, this loss of pressure will have to be viewed at the HOPM during the initial system investigation.

Loss of Hydraulic Oil Pressure

Usually, a loss of hydraulic oil pressure will cause an alarm to be generated at the PACC/PCC. The generation of this alarm, of course, will immediately alert the operator to a problem. The alarm will sound when casualties occur either to the main relief valve or to a sequencing valve. The alarm may or may not sound, however, when a major leak occurs. Look at the three main causes of hydraulic oil pressure loss and the resulting alarms.

MAIN RELIEF VALVE.— A faulty main relief valve can be identified by a low-pressure alarm at the console, but the actual answers can be found at the HOPM. This component failure can be easily identified by the operator. The operator simply looks at the HOPM pressure gauges and notes that all pressures are extremely low or nonexistent.

SEQUENCING VALVE.— You may suspect that a sequencing valve is faulty after a loss of pitch control as well as a loss of hydraulic oil pressure, depending on how the valve fails. For instance, if the valve fails in the open position, then all the oil would become high-pressure oil and the low-pressure alarm would not sound. In this case, the auxiliary servo supply (control oil) pressure would be drastically low and the system control would fade. On the other hand, if the sequencing valve fails in the closed position, a low-pressure alarm would sound and alert the operator. In this instance, the operator would also be able to see an extremely sluggish pitch response time.

MAJOR LEAK.— A major leak can provide the same symptoms as a loss of hydraulic oil pressure, depending on the location of the leak.

We have just told you about some of the components you should check when you detect a loss of pitch control or a loss of hydraulic oil pressure. Now, we will discuss some of the most common maintenance procedures you as a supervisor will expect your personnel to perform.

Cleaning and Lubrication

Cleaning is a continuous task. As a GS supervisor, you are already aware that good housekeeping practices must be maintained and passed on to your subordinates. The responsibilities for cleaning and lubricating the components of the CRP/CPP systems are very similar to those for the MRG LO system. This is because the majority of the components that require cleaning in both the MRG LO system and the CRP/CPP systems are valves. When cleaning the valves of the CRP/CPP systems, be sure to pay close attention to detail. Attention to detail is important because most of the valves and piping of the CRP/CPP systems are located in the bilge area. Valves in the bilge area are constantly exposed to corrosive elements.

Other components that require cleaning and lubrication are the two CRP/CPP pump couplings. These pump couplings are not cleaned and lubricated as frequently as the valves, but their cleaning and lubricating are still very important responsibilities.

Alignments and Adjustments

You will routinely supervise alignments and adjustments to couplings and other system components. Your personnel usually perform these tasks after general maintenance (cleaning and lubrication). Alignments and adjustments are either scheduled or conditional. During the cleaning process, for example, you may discover that a coupling requires an alignment check or adjustment.

The CRP/CPP system is one of the few systems that you as a supervisor will be required to train your personnel to closely monitor locally. Local monitoring is necessary because of the lack of remote monitoring capabilities. You will also be required to train your personnel to make the necessary mechanical and electrical adjustments. Your personnel will periodically perform these procedures through your ship's PMS. Remember, first you must monitor the operation of the CRP/CPP system as a whole, and then isolate individual components (one at a time) to ensure they are functioning properly. The following paragraphs contain some of the components you maybe required to adjust and the functions they are designed to perform.

UNLOADING VALVE.— The unloading valve unloads the pressure of the attached pump back to the sump if the electric pump is operating and functioning properly.

SEQUENCING VALVE.— The sequencing valve serves two purposes:

(1) It maintains a back pressure on the system to ensure that a minimum of 400 psi is supplied to the inlet side of the reducing valve, and (2) it provides high-pressure oil to the OD box.

REDUCING VALVE.— The reducing valve provides control oil to the OD box.

AUXILIARY SERVO RELIEF VALVE.— The auxiliary servo relief valve relieves excess control oil pressure back to the sump.

MAIN RELIEF VALVE.— The main relief valve relieves excessive pump pressure, either from the electric pump or attached pump, back to the sump.

Besides adjusting the components at the OD box, both mechanical and electronic pitch position alignment checks must be accomplished periodically. These checks will not only require your expertise to train your personnel, but also require your presence while they are being accomplished.

MECHANICAL ALIGNMENT.— The mechanical alignment procedure is basically the same for all the ship classes. This procedure is performed according to the PMS and is used to detect valve rod separation (unscrewing) or elongation. Remember, two people will be required to perform this check One must be positioned at the OD box and the other at the HOPM, and they must be able to communicate with each other (sound-powered phones or walkie-talkies). This test is normally fairly easy to accomplish if no problems are encountered. By problems we mean the pitch scale and the pitch position pointer being off by more than 1/16 of an inch. If this difference cannot be explained by thermal growth or contraction of the valve rod assembly, it will be necessary to verify that all connections in the valve rod assembly are tight. If the position of the pointer and pitch scale is subject to question at anytime, you must verify actual position of blade 1A to the hub body marks. If the ship is not in dry dock, you must use a diver to observe and confirm hub body marks. You must have confirmation of the hub body marks at design ahead and full ahead when pitch is ordered at normal operating temperatures.

NOTE

In most cases, an equipment malfunction is not the cause of the pointer and scale discrepancy. Usually, it is an operator error. To avoid this problem, make sure your personnel strictly follow the MRC and always take all measurements at the same system oil temperature.

On the other hand, if the pitch pointer and the scale difference is less than a 1/16 of an inch or the greater than reading is related to thermal growth or contraction, then the scale can be moved to match the pointer's position. The only drawback to adjusting the scale to match the pointer is that an electronic alignment (calibration) must be performed

ELECTRONIC ALIGNMENT.— The electronic alignment procedures differ depending on the ship class. But, one thing will always be the same. To accomplish any of these procedures, the ship must be in dry dock or you will require the assistance of a diver. All adjustments made to align the mechanical (actual) pitch and electronic display indications must be verified with the actual blade position on the propeller hub. There is one electronic alignment procedure (electronic pitch indicating [EPI] system calibration) on the DDG-51 class that can only be accomplished when the ship is in dry dock.

REMOVAL AND REPLACEMENT OF COMPONENTS

The CRP/CPP system seldom requires the removal or replacement of components. However, there is one component that you will be required to replace—the electrohydraulic servo valve. As previously discussed, this valve is in constant use and its probability of failure is much higher than any other component in the system.

There is only one other set of components that you will need to remove frequently—the system's filters. In fact, you will need to remove these filters even more frequently than the electrohydraulic servo valve. This is because you will need to remove the filters for periodic cleaning according to the PMS. Of course, you will also need to remove them in the event of a casualty.

SUMMARY

In this chapter, we have discussed many of the factors that affect GTE performance, power train operation and maintenance, and propulsion systems. As you prepare for advancement, you must continue to learn and increase both your leadership skills and technical expertise. You must be prepared to train and supervise your subordinates. Read the various reference materials cited in this chapter to increase your understanding of the information that was presented.